I. Amendments to the Specification

Kindly replace paragraph number [0007], on page 2, with the following rewritten paragraph:

[0007] Within the condenser 14, the superheated refrigerant fluid changes from its gaseous phase to a mostly liquid phase. The superheated vapor of the refrigerant fluid flows through interior passages 30 of the condenser 14 while ambient air flows over exterior surfaces 32 and cooling fins 34 of the condenser 14. The superheated vapor is much hotter than the ambient air. Thus, the heat of the superheated vapor is given off to the surrounding ambient air flowing over the exterior surfaces 32 and cooling fins 34 of the condenser 14, thereby cooling the refrigerant fluid in accord with heat transfer principles. As the refrigerant fluid continues to flow through the condenser 14 and lose more heat to the surrounding ambient air, it begins to condense from its gaseous phase into a liquid phase. Eventually, the refrigerant fluid exits the condenser 14, mostly in a liquid phase (X) but typically including some gaseous portion, and flows downstream through the refrigerant line 21, and enters the receiver-dryer 22.

Kindly replace paragraph number [0012], on page 4, with the following rewritten paragraph:

[0012] In prior art air-conditioning systems, under vehicle usage conditions there may - or may not- be sub-cooling at the output side (range X – in Figure 6 ~~5~~, B-C) of the condenser (14 in Figure 5), depending upon the state of the refrigerant fluid due to various vehicle performance variables. In other words, and referring to Figure 6, range X represents the variable nature of the refrigerant fluid temperature at the downstream or output side of the condenser 14 at range X in Figure 5 and Y_1 represents the sub-cooling of prior art refrigeration cycle. Whereas point A is well defined and fixed at the location on the pressure vs. enthalpy diagram as shown, range X is

not so well defined and varies along the condenser path A-D of the pressure vs. enthalpy diagram depending upon the vehicle performance variables of vehicle speed and load on the air-conditioning system. The slower the vehicle speed, or at idle condition and, the higher the load on the air-conditioning system, the sub-cooling range Y_1 diminishes and may approach zero. Under these conditions, the refrigeration cycle looses sub-cooling capability and operates only in the "X" range. Likewise, point D is dependent upon the amount of sub-cooling that can be performed on the refrigerant beyond point C. In other words, point D is incrementally dependent upon the cooling load and quantity of ambient air flow when the air conditioning system is properly charged with refrigerant.

Kindly replace paragraph number [0036], on page 12, with the following rewritten paragraph:

[0036] The compressor 112 is mounted within an engine compartment of a motor vehicle (not shown) such that the compressor 112 is powered by an accessory drive belt 126 that connects to a crankshaft pulley of an engine (not shown) or is electrically driven (not shown). Rotation of the engine translates into rotation of the compressor pulley to power the compressor 112 when a clutch ~~126~~ <u>128</u> on the compressor 112 is engaged. Accordingly, the compressor 112 suctions gaseous refrigerant from an upstream portion of the refrigerant line 124" into an inlet port 130 thereof, compresses the gaseous refrigerant into a high pressure, high temperature superheated gaseous state, and pumps the refrigerant out an outlet 132 downstream toward the IRDC 114. Referring to the pressure vs. enthalpy diagram of Figure 2, this compression process is represented by path O-A.

Kindly replace paragraph number [0037], on page 12, with the following rewritten paragraph:

[0037] Referring again to Figure 1, the condenser 116 of the IRDC 114 generally includes a pair of opposed header tanks defined by a first header tank 134 and a second header tank 136, and further includes a heat exchanging core 138 fluidically connected between the header tanks 134, 136. The heat exchanging core 138 includes a plurality of horizontal tubes or passages 140 having opposed ends in fluidic communication with the header tanks 134, 136. Corrugated cooling fins 142 are disposed between exterior surfaces 144 of the passages 140 for cooling the refrigerant flowing therethrough. The header tanks 134, 136 are basically vertically disposed hollow vessels having horizontal partitions, dividers, or separators D1-D5 therein. The first header tank 134 includes an inlet port 146 and the opposite, second header tank 136 includes an outlet port 148. It is contemplated, however, that one or the other of the header tanks 134, 136 could include both the inlet port and outlet ~~ports~~ port 146, 148 depending upon how many and in what location the horizontal partitions D1-D5 are used. Thus far described, the condenser 116 is preferably composed of aluminum, is manufactured in accordance with known condenser manufacturing techniques, and is designed in accord with typical condenser design configurations, with the below-mentioned exceptions.

Kindly replace paragraph number [0038], on page 13, with the following rewritten paragraph:

[0038] Preferably, five separators D1, D2, D3, D4, D5 are used to divide the condenser 116 into sub-sections. A condensing stage of the condenser 116 is defined between the inlet port 146 and the fifth separator D5, and a sub-cooling stage is defined between the fifth separator D5 and the outlet port 148. The fourth and fifth separators D4, D5 are disposed at the same elevation within their respective header tanks 136, 134, such that there is no fluidic

communication between the condensing and sub-cooling stages within the condenser 116 itself. A person skilled in the art will recognize that the number of separators used is a function of the application and therefore the five separators D1-D5 as disclosed in the preferred embodiment is not intended to be limiting. Any number may be used, or adapted for the application.

Kindly replace paragraph number [0039], on page 14, with the following rewritten paragraph:

[0039] However, the receiver-dryer 118 of the IRDC 114 fluidically communicates the condensing stage of the condenser 116 to the sub-cooling stage of the condenser 116. The receiver-dryer 118 communicates with an intermediate outlet port 150 at the end of the condensing stage of the condenser 116 via an inlet tube, stand pipe, line 152, or the like, that extends centrally and upwardly within a generally cylindrical housing 154 and terminates in an exit end 156 in an upper portion 158 of the cylindrical housing 154. An integrated filter and adsorbent unit 160 is mounted about the inlet line 152 for dehydrating or removing water from the refrigerant. An outlet line or tube 162 extends downwardly from a lower portion 164 of the cylindrical housing 154 and communicates through an intermediate inlet port 166 with the sub-cooling stage of the condenser 116. The inlet and outlet lines 152, 162 are preferably brazed or joined mechanically to the cylindrical housing 154 and connected to the condenser 116 using tube connecting blocks (not shown), which are known in the art. The receiver-dryer 118 is shown positioned beside the condenser 116, but may be positioned in front thereof to maximize the efficiency of the refrigerant by using cooling fins 175 as shown in Figure 3. The unique design and construction of the receiver-dryer 118 will be discussed in more detail below with regard to Figures 3 and 4.

Kindly replace paragraph number [0040], on page 14, with the following rewritten paragraph:

[0040] The following discussion will refer simultaneously to the apparatus of Figure 1 and to the graphical depiction of the function of that apparatus in Figure 2. Referring to Figure 1, the refrigeration cycle continues within the IRDC 114 to change the pressurized refrigerant fluid from its gaseous phase to a liquid phase, as represented by path A-D' in the pressure vs. enthalpy diagram of Figure 2. Referring to Figure 1, the superheated vapor of the refrigerant fluid flows back and forth, winding its way down through the interior of the passages 140 of the condenser 116 while ambient air flows over the cooling fins 142 and exterior surfaces 144 of the passages 140. The superheated vapor is much hotter than the ambient air and, thus, the heat of the superheated vapor is given off to the surrounding ambient air flowing over the <u>cooling</u> fins 142 and other exterior surfaces 144 of the condenser 116, thereby cooling the refrigerant fluid in accord with heat transfer principles. In other words, as the superheated vapor of the refrigerant fluid continues to flow through the condenser 116 and lose more heat to the surrounding ambient air, it begins to condense from its high pressure superheated gaseous phase into a high pressure liquid phase. Point B in the pressure vs. enthalpy diagram of Figure 2 corresponds to a location in the condenser 116 of Figure 1 that likely varies between the inlet port 146 and the second separator D2.

Kindly replace paragraph number [0041], on page 15, with the following rewritten paragraph:

[0041] Similar to prior art Figures 5 and 6, point X of Figure 1 corresponds to the variable range X depicted in Figure 2, wherein the refrigerant exiting the intermediate outlet port 150 is predominantly a liquid phase but also includes some gaseous phase as a result of the cooling capacity. Like the previous discussion with reference to Figure 6, here range X in Figure

2 represents the liquid and gaseous phase of the refrigerant fluid at an intermediate portion of the condenser 116 at point X in Figure 1. Whereas point A in Figure 2 is well defined and fixed at the location on the pressure vs. enthalpy diagram as shown, any one point within range X is not so well-defined and varies along the condenser path B-C (146 to 150 and from 166-148) of the pressure vs. enthalpy diagram depending upon the vehicle performance variables of vehicle speed and load on the air-conditioning system as illustrated in Figure 1 from reference character 146 to 150 and 166 to ~~144~~ 148'. The slower the vehicle speed and at idle, and the higher the load on the air-conditioning system, any one point within the range X will move in the direction of point B. In other words, as can be seen in Figure 2, the point within range X can vary from a saturated vapor to a sub-cooled liquid or anywhere in between such as a liquid-vapor mixture. In contrast to the prior art system and diagram of Figures 5 and 6, here with the system and diagram of Figures 1 and 2 of the present invention, point D' is providing additional amounts of sub-cooling that can be performed within the system Y_2.

Kindly replace paragraph number [0042], on page 16, with the following rewritten paragraph:

[0042] Rather, point D' is also influenced by the ability of the present invention to provide subsequent efficient sub-cooling and separation of liquid and gas phases of the refrigerant fluid beyond point $X+Y_1$ (between ~~point~~ range X and ~~point~~ range Y_1) and further subsequent sub-cooling beyond ~~point~~ range Y_1 to ~~point~~ range Y_2. As shown in Figure 1, the receiver-dryer 118 is a vertically disposed vessel for separating the refrigerant wherein the mixture of gaseous-liquid phase rises to the top of, and captures the gaseous phase within the upper portion 158 thereof, yet the liquid phase of the refrigerant falls under gravity and settles in the lower portion 164 thereof. Accordingly, location Y in Figure 1 corresponds to the sub-

cooling range Y_1+Y_2 depicted in Figure 2, wherein the refrigerant entering the intermediate inlet port 166 of the condenser 116 is saturated or sub-cooled liquid refrigerant (point C). The refrigerant at location C is mostly saturated liquid refrigerant at location X, because the refrigerant at location X is a varying combination of liquid and gaseous phases whereas the refrigerant at location Y (166) is a stable supply of liquid phase separated in the bottom chamber or outlet line 162 of the lower portion 164 of the receiver dryer ~~164~~ 118. Additional sub-cooling takes place within the condenser 116 between the intermediate inlet port ~~or point~~ 166 and the outlet port 148 whereat the pressurized sub-cooled refrigerant fluid exits the condenser 116 at Point D' as a liquid phase, flows downstream through the refrigerant line 124 ~~124'~~, and enters the thermal expansion valve 120.

Kindly replace paragraph number [0043], on page 17, with the following rewritten paragraph:

[0043] Accordingly, the present invention ensures the presence of sub-cooling and increases the magnitude thereof. This can best be seen by comparing the leftward shift of line D'-F' of Figure 2 compared to the position of line D-F of prior art Figure 6. In other words, the present invention increases the enthalpy range from point ~~O~~ F' to point ~~F'~~ O as seen in Figure 2, compared to the prior art enthalpy range from point F ~~O~~ to point ~~F~~ O of Figure 6. The amount of heat (Q) that can be removed by the present invention air-conditioning system is represented by the equation $Q=M_{R134a}*(h2-h1')$. M_{R134a} is the variable mass flow for R134a refrigerant while h2 is the enthalpy at the end of the compression cycle O-A and h1' is the enthalpy at the end of the condensing cycle A-D'. Assuming a constant mass flow, the greater the range in enthalpy that the air-conditioning system can produce, the greater the heat that can be removed. Therefore, by increasing the enthalpy range compared to the prior art, the present invention thereby increases

the amount of heat that can be removed from the refrigerant fluid, which translates to an increase in efficiency of the present invention air-conditioning system compared to the prior art.

Kindly replace paragraph number [0046], on page 19, with the following rewritten paragraph:

[0046] Figure 3 illustrates an enlarged view of the receiver-dryer 118 shown in Figure 1. The cylindrical housing 154 is preferably composed of a thin-walled metal such as a 6063-T6 aluminum alloy, but may be composed of other aluminum, steel, plastic, and the like. The inlet and outlet tubes 152, 162 are preferably brazed to the cylindrical housing 154 and are preferably composed of a 3003-H14 aluminum alloy, but may be composed of other aluminum, steel, plastic, and the like. The receiver-dryer 118 of Figure 1 is a substantially cylindrical vessel, container, or housing having a base wall 170, a side wall 172 extending vertically upwardly from the base wall 170, and a concave end 174 terminating the side wall 172. The concave end 174 need not, but may, take the form of a thin-walled spherical wall, just as long as a concave interior surface is defined by the concave end 174. The walls 170, 172, 174 collectively define an interior of the ~~receiver-dryer~~ cylindrical housing 154. The refrigerant inlet pipe 152 extends into the interior of the cylindrical housing 154 and terminates in the exit end 156 facing the concave interior surface of the concave wall 174 of the cylindrical housing 154. The receiver-dryer 118 also includes the integrated filter and adsorbent unit 160 that is centrally disposed over the inlet tube 152 and that is elevated by one or more indentations 176 formed into the side wall 172 of the cylindrical housing 154. The adsorbent unit 160 may be a saddle bag type device, a puck-like device, or any other suitable desiccant and filter device that is known. The adsorbent unit 160 effectively divides the interior of the cylindrical housing 154 into the upper portion 158 above the adsorbent unit 160 and the lower portion 164 below the adsorbent unit 160.

Kindly replace paragraph number [0047], on page 20, with the following rewritten paragraph:

[0047] The inlet tube 152 is adapted for directing the refrigerant fluid into contact with the concave end wall 174 such that the refrigerant fluid impinges on the inner concave end wall 174 to separate the mixture of liquid/gaseous refrigerant fluid into a gaseous phase that accumulates in the upper portion 158 of the cylindrical housing 154 and a liquid phase that by adhering to the interior concave end wall 174 falls under gravity to accumulate in the lower portion 164 of the cylindrical housing 154. The design of the concave wall 174 and proximity of the exit end 156 of the inlet tube 152 is adapted for substantial contact of liquid refrigerant and relatively uniform dispersion of refrigerant so that a substantial amount of refrigerant liquid adheres to the inner surfaces of the cylindrical housing 154 due to liquid surface tension and wherein the liquid runs down interior surfaces of the concave wall 174 and side wall 172 for heat transfer cooling therewith. Additional efficiency maybe obtained by the use of cooling fins 178 as shown in Figure 3. Therefore, cooling fins 178 are preferably disposed on the exterior of the cylindrical housing 154 for increased heat transfer cooling of the refrigerant fluid. The combined secondary surface area of the cooling fins 178 is represented by element A_S and the combined primary surface area of the concave wall 174 and side wall 172 in the upper portion ~~168~~ 158 of the cylindrical housing 154 is represented by element A_p. According to the present invention, A_S is preferably greater than A_p. The unique design of the concave wall 174 and proximity of the inlet tube 152 with respect thereto enables relatively greater dispersion of the refrigerant fluid, and the cooling fins 178 enable relatively greater conversion of the refrigerant fluid into a liquid phase. Both features provide for greater condensing of the refrigerant gas phase into liquid phase. The cooling fins 178 may be separately attached to the cylindrical housing 154 such as by

brazing, or may be assembled thereto as a separate sub-assembly. In a similar vein, Figure 4 illustrates an alternative embodiment of the present invention, in which the heat transfer functionality of the cooling fins 178 (shown in Figure 3) is substituted by an isomount hat 180 or maybe integrated with the cooling fins 178 (shown in Figure 3).

Kindly replace paragraph number [0048], on page 21, with the following rewritten paragraph:

[0048] The isomount hat 180 includes a socket shaped portion 182 that is adapted for heat transfer contact with the top of the cylindrical housing 154 and further includes a bracket portion 184 that is adapted for fastening to another structural member such as the condenser 116 or any other proximate structure within an engine compartment. Accordingly, the top of the receiver-dryer 118 may be firmly supported and mounted within the engine compartment for less vertical and lateral movement of the receiver-dryer 118. The socket shaped portion 182 is concave shaped for conforming contact with the convex shaped concave wall 174 of the cylindrical housing 154. The socket shaped portion 182 is also preferably constructed of a relatively high thermally conductive material such as aluminum or steel and may have a metallic or non-metallic outer skin. It is contemplated that the isomount hat 180 could be used in combination with the cooling fin 178 arrangement of Figure 3. In any case, a secondary surface area As' should be greater than the primary surface area Ap.

Kindly replace paragraph number [0049], on page 21, with the following rewritten paragraph:

[0049] Referring again to Figure 3, the outlet ~~tube~~ line 162 has an entrance end 186 in fluidic communication with the lower portion 164 of the cylindrical housing 154 for permitting only the liquid phase of the refrigerant and a lubricant to exit the receiver-dryer 118. The level of

saturated liquid and lubricant will change depending upon the condensing capacity of the apparatus, the cooling load placed on the refrigeration system, vehicle performance, and the like.

Kindly replace paragraph number [0050], on page 22, with the following rewritten paragraph:

[0050] The receiver-dryer 118 may be manufactured according to any of the well-known techniques for forming aluminum canisters, but is preferably constructed by the following described process. The <u>cylindrical</u> housing 154 preferably originates from tube stock which is impact closed to form the flat bottom end or base wall 170. However, the <u>cylindrical</u> housing 154 may originate from sheet or tubular stock, which is then deep drawn to form the base wall 170. Holes are then drilled in the closed bottom end or base wall 170 and the inlet and outlet tubes 152, 162 are inserted therein and brazed to the <u>cylindrical</u> housing 154. The inlet tube 152 is inserted within the <u>cylindrical</u> housing 154 such that the exit end 156 thereof faces the top inside surface of the concave wall 174 <u>after formation thereof</u> and is disposed within a distance that is substantially proximate the radius of the spherical-shaped concave wall 174 of the <u>cylindrical</u> housing 154. Alternatively, the exit end 156 may be spaced from the top inside surface within proximity of the radius dimension of the spherical concave wall 174. Then, the indentation(s) 176 are formed in the side wall 172 of the <u>cylindrical</u> housing 154 by tri-crimping or forming cylindrically the <u>cylindrical</u> housing 154, or the like. Next, the integrated filter and adsorbent unit 160 is assembled into the interior of the <u>cylindrical</u> housing 154. The open end of the tube stock is spun closed to form the closed top end or concave interior wall 174. Spin closing of aluminum containers is generally known in the art, e.g. by U.S. Pat. No. 5,245,842, which is incorporated by reference herein. Uniquely, however, the top end or concave wall 174

is preferably spun closed in such a manner so as to achieve a concave, rounded, and preferably spherical, top inside surface of the concave wall 174.

Kindly replace paragraph number [0051], on page 22, with the following rewritten paragraph:

[0051] In accordance with the present invention, the preferred method involves improved sub-cooling of the refrigerant within an air conditioning system. The method may be practiced in accord with the ~~air conditioning~~ refrigeration system 110 of Figure 1, but may also be practiced using any suitable air conditioning system. The method includes receiving a superheated gaseous phase of a refrigerant fluid in a condensing stage of a condenser, and condensing the superheated gaseous phase of the refrigerant fluid within the condensing stage into a mixture of a gaseous phase and a liquid phase of refrigerant. The method further involves communicating the mixture into a vertically disposed container, housing, or vessel, and directing the mixture into a top concave surface of the vertically disposed container, thereby dispersing the liquid phase from the gaseous phase wherein the liquid phase falls toward a lower portion of the container over a desiccant material, and further thereby cooling the gas and liquid phases for improved sub-cooling of the liquid phase by adhering to the interior concave wall 174 and for improved condensing of the gas phase into the liquid phase. Accordingly, the method produces a separated, cooled, and dehydrated liquid phase that accumulates in the lower portion of the container. Finally, the method includes communicating the separated, cooled, and dehydrated liquid phase out of the container and back into a sub-cooling stage of the condenser.